Journal of NUCLEAR SCIENCE and TECHNOLOGY, Vol. 41, No. 7, p. 765–770 (July 2004)

**TECHNICAL REPORT**

# Experimental and Operational Verification of the HTR-10 Once-Through Steam Generator (SG)

## Heat-transfer and Hydrodynamic Features of the Once-Through Steam Generator

Huaiming JU*, Youjie ZHANG, Zhiyong HUANG, Zhiyong LIU, Jun LI and Yu YU

*Institute of Nuclear Energy Technology, Tsinghua University, Beijing 100084, China*

(Received August 18, 2003 and accepted in revised form April 21, 2004)

In January 2003, the 10 MW High-temperature Gas-cooled Reactor (HTR-10) reached its full power for continuous operation of seventy-two hours in the Institute of Nuclear Energy Technology, Tsinghua University. The reactor was operated smoothly at the designated parameters. The once-through steam generator (SG) is one of key equipments of the HTR-10 reactor. The SG includes 30 modular heating helical tube assemblies. Design of the SG includes hydraulics, heat transfer and stability designs. Based on the design requirement, it is necessary to ensure sufficient heat removal from the reactor in order to maintain stable operation. In order to confirm the thermal hydraulic reliability of the SG, a series of experiments had been carried out. The purpose of this paper is to introduce the design features and experimental verification of HTR-10 SG, and the research results of small bending radius helical coil-pipe used in HTR-10, for example, the heat transfer coefficient of water, superheat steam and the two phase flow in the helical tube, the heat transfer coefficient of the He flow across the helical tube, and the centrifugal force effect on the heat transfer for the helical tube. In the paper, some operational experimental data of the HTR-10 SG have been presented.

***KEYWORDS: HTGR type reactors, once-through steam generator, helical tube steam generator, heat transfer***

## I. Introduction

In January 2003, the 10 MW High Temperature Gas-cooled Reactor (HTR-10) reached its full power for continuous operation of seventy-two hours in the Institute of Nuclear Energy Technology, Tsinghua University. The reactor was operated stably and the operational performance was satisfactory.

Once through steam generator (SG) is one of key equipments of the HTR-10. The modular helical tube assemblies is used in the HTR-10 SG. The SG consists of 30 mono-spiral helical tubes.[1)] Every helical tube is a modular and its structure parameters were presented in **Table 1**.

Under effects of the gravitational force, centrifugal force and secondary flow there are some differences between the straight and the helical tubes. Flow in a helical tube has obvious asymmetry about the flow axis. These effects are dependent on the curvature of the helical tube and the velocity of the coolant and quality.

So far some experimental study on the heat transfer of the helical tube has been performed by different authors.[1,2)] The range of the diameter ratio between the curvature diameter of the helical tube and the diameter of the tube ($D/d$) in these experiments is about 11 to 65. Experimental results showed that in the helical tube the distribution of wall temperature and heat transfer coefficient are asymmetrical about the flow axis. The smaller the diameter ratio is, the stronger the effect of the centrifugal force becomes and more asymmetrical the flow becomes. Though some calculation formulas of the heat transfer coefficient have been given, these formulas are applicable only within a limited range of the diameter ratio.

**Table 1** Structure parameters of the HTR-10 SG

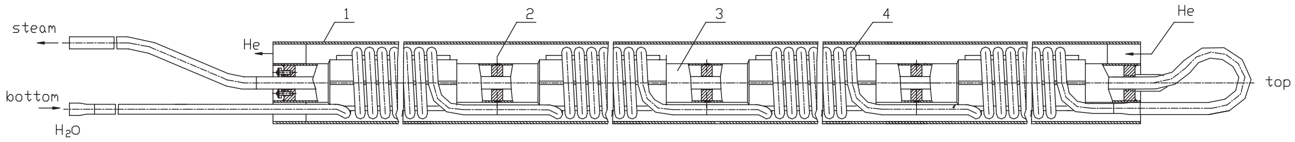
| | |
|---|---|
| Diameter of the heat transfer tube | 18 mm |
| Curvature diameter of the helical tube | 112 mm |
| Vertical pitch of the helical tube | 22.5 mm |
| Diameter of the center tube | 84 mm |
| Inner diameter of the casing tube | 140 mm |
| Length of the heat transfer tube in every ring | 0.3526 m |
| The climb angle of the helical tube | 3.66° |
| Number of the heat transfer tube | 30 |
| Diameter of the heat transfer tube (water section , nuclear boiling) | $\Phi 18\times 3$ mm |
| Diameter of the heat transfer tube (superheating steam section) | $\Phi 18\times 2$ mm |

Diameter ratio of the helical tube for the HTR-10 SG is small. ($D/d\cong 9$). The empirical formulas in publications can be used for reference are very limited, so special experimental study is required.

## II. Design Features of the HTR-10 SG

The small helical tube assemblies is used in the HTR-10 SG. Assembly is shown in **Fig. 1**. Helium flows in an annular cavity between the central tube and the casing tube. High temperature helium enters from the top of SG, across the helical tube and flows downward. Sub-cooled water enters from the bottom of helical tubes, and then flows upward in the helical tubes. After heat exchange between two mediums, high temperature helium is cooled and then leaves the helical

*Corresponding author, Tel.+86-10-62784833, Fax.+86-10-62771150, E-mail: juhm@mail.tsinghua.edu.cn




1: Casing tube, 2: Annular cavity, 3: Central tube, 4: Helical tube

**Fig. 1** Structural scheme of SG modular assembly

tubes, while sub-cooled water turns into superheated steam and also returns to the bottom of test assembly through the down-comer within the central tube. The SG consists of 30 modular assemblies. The assemblies are arranged in a pressure vessel.

The main advantages of the assembly structure are: simplicity in the manufacture, convenience for the service and repair, and compact structure. The structure parameters of the HTR-10 SG are shown in Table 1.

From Table 1 can be seen that the diameter ratio of the heat transfer helical tube is small (only 9.3 and 8). In the heat transfer design, the heat transfer tube is divided into single-phase water preheating section, nucleate boiling section, film boiling section and superheated steam section. The heat transfer calculation was performed on the base of straight tube heat transfer calculation with an additional correction of the helical tube. The correctional coefficients was determined by experiments. The flow are divided into Laminar flow regime, laminar flow with eddy regime, turbulent flow regime and auto-modeling regime. In judging the flow patterns the effect of the helical curvature was considered. The De number was introduced in the calculation,

$$De = Re\sqrt{d/D},$$

where $De$ is the Dean number, $Re$ is the Reynolds number, $d$ is the diameter of the heat transfer tube and $D$ is the curvature diameter of the helical tube.

In the primary design, the following formulas[1)] were used:

$$Re_{cr} = 2,100\left[1 + 12\left(\frac{d_{\rm in}}{D}\right)^{0.5}\right], \tag{1}$$

where $Re_{cr}$ is the transition critical $Re$ from laminar to turbulent flow.

In the primary design, correction coefficients of the helical tube are as follows:[3)]

$$\text{In the laminar regime,} \quad \frac{Nu}{Nu_{st}} = 1 \tag{2}$$

In the turbulent regime,

$$\frac{Nu}{Nu_{st}} = 1 + 6.3\left(1 - \frac{d_{\rm in}}{D}\right)\left(\frac{d_{\rm in}}{D}\right)^{1.15}, \tag{3}$$

where $Nu_{st}$ is the Nusselt number for straight tube, $Nu$ is the Nusselt number for helical tube.

The heat transfer coefficient is a function of the $Re$, and vertical pitch of the helical tube $S$.

After a long period of operation, some corrosion products may deposit on the heat transfer surface. It causes an increase of the heat resistance. So in the heat transfer design heat transfer surface must has a sufficient amount of the design margin. A fouling factor must be considered.

## III. Experimental Verification of the HTR-10 SG

In order to study and verify the thermal-hydraulic performance of the SG, a full-scale HTR-10 Steam Generator Two Tube Engineering Model Test Facility (STGM-10) was installed. The system flow chart has been shown in **Fig. 2**. The test assembly of the STGM-10 simulates practical thermal and structural parameters of the HTR-10. The SGTM-10 consists of three separated loops: primary helium loop, secondary water loop and third cooling water loop. There are two parallel tubes arranged in the test assembly. The coolant flow rate, pressure, pressure drop, temperatures were measured according to the experimental requirements. Besides the coolant temperature measurement, there are 28 groups of tube wall temperature measurement points. The arrangement of measurement points in the across section of the tube have been shown in **Fig. 3**.

The main error of measurement are as follows: the temperature measurement error is 0.65 to 2.8°C (when $t \geqslant 450$°C); the flow rate measurement error is $\pm 1.1\%$; the pressure drop and pressure measurement error is $\pm 0.15\%$; the power measurement maximum error is $\pm 2\%$.

A series of experiments had been carried out on the test loop STGM, such as studies of on two-phase flow stability, hydraulic characteristics.[4–6)]

The heat transfer study for the small helical tubes is presented in the paper.

### 1. Verification of Heat Transfer Coefficient for Water and Steam

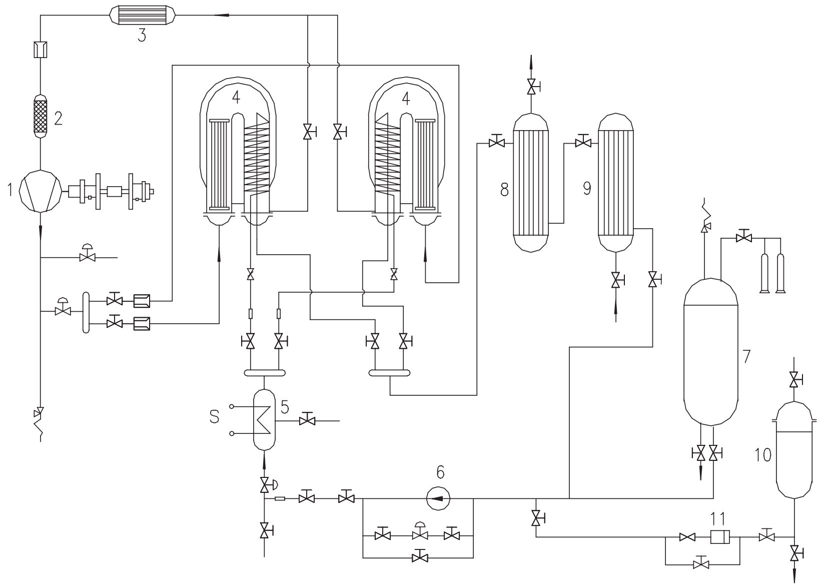
In the SG model, the temperatures of He, tube wall temperature, water and steam temperature, water and steam flow rate, and pressure of water and steam were measured. Among these parameters there are following relationships:

$$q_1 = \alpha_{\rm He}(t_{\rm He} - t_{\rm w}) \tag{4}$$

$$q_1 = K(t_{\rm He} - t_{\rm H_2O}) \tag{5}$$

$$K = \left(\frac{d_{\rm out}}{d_{\rm in}} \cdot \frac{1}{\alpha_{\rm H_2O}} + R_{\rm f} + R_{\rm w} + \frac{1}{\alpha_{\rm He}}\right)^{-1}, \tag{6}$$

where $q_1$: Heat flux density (kW/m)
$\alpha_{\rm He}$: Heat transfer coefficient from the He to the wall (kJ/m·°C)
$\alpha_{\rm H_2O}$: Heat transfer coefficient from the wall to the water (kJ/m·°C)




1. Helium circulator, 2. Filter, 3. Cooler, 4. Electric heater and steam generator, 5. Pre-heater, 6. Canned pump, 7. Pressurizer, 8. Cooler, 9. Heat exchanger, 10. Water tank, 11. Plunger pump

**Fig. 2** System flow chart

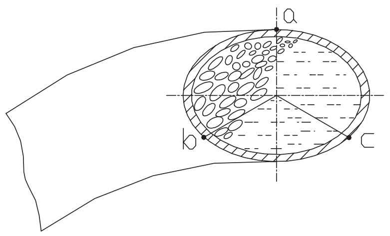



**Fig. 3** Flow model in the section of a helical pipe

K: Total heat transfer coefficient (kJ/m·°C)
$t_{\mathrm{He}}$: Temperature of the He (°C)
$d_{\mathrm{in}}$: Inner diameter of the heat transfer tube
$d_{\mathrm{out}}$: Outer diameter of the tube
$t_{\mathrm{H_2O}}$: Temperature of the water or steam (°C)
$t_{\mathrm{w}}$: Outer tube wall temperature (°C)
$R_{\mathrm{w}}$: Thermal resistance of the tube wall conductivity
$R_{\mathrm{f}}$: Fouling thermal resistance.

From these relationships and measured experimental data, the heat transfer coefficients of the sub-cooling water and the superheated steam in the SG can be obtained. Experimental results verified the calculation formulas used in the design. It proved that the calculation formula (3) is coincident with experimental data. The variances are shown in **Figs. 4** and **5**. In Figs. 4 and 5, the heat transfer coefficients are represented by wall temperatures. Where $t_{w,ex}$ is the measured wall temperature, $t_{w,ca}$ is the calculated wall temperature, $\alpha_{ex}$ is the measured heat transfer coefficient and $\alpha_{ca}$ is the calculated heat transfer coefficient. The variance is about ±5%.

## 2. Heat Transfer of the Two-Phase Flow in the Helical Tube

The thermal resistance in the once-through helical tube SG is mainly caused by the He-gas heat transfer process. In the verification of the two phase flow heat transfer coefficient in the helical tube, the combination method of the theoretical calculation with experiment is used. In the verification, 28 groups of measured wall temperatures and a steady-state analysis computer code were used.

The comparison between the experiment and analysis is shown in **Fig. 6**.

From Fig. 6 can be seen that for different quality the measured wall temperatures basically are agreeable to the calculated wall temperatures.

The variance is basically no more than ±5%.

## 3. Heat Transfer Coefficient of the He Flow Across the Helical Tube

As mentioned above, coat tube type arrangement is used in the assembly of the HTR-10 SG. The He in primary side flows down between the outside casing tube and the center tube across the helical tube assembly. Its heat transfer coefficient depends on the ratio of the vertical pitch of the helical tube to the tube diameter. In our case, the ratio of the vertical pitch of the helical tube $S$ to the diameter of the helical tube d, $\delta = S/d = 1.2$. From experimental data and Eqs. (4) through (6), the following formula was obtained:

$$Nu_{\mathrm{He}} = 1.15 Re^{0.65} Pr^{0.33}. \tag{7}$$

This formula can be applied only for given $\delta$ ($\delta = S/d_{out} = 1.2$).

## 4. Verification in Operation

In January 2003, the HTR-10 was operated with full pow-

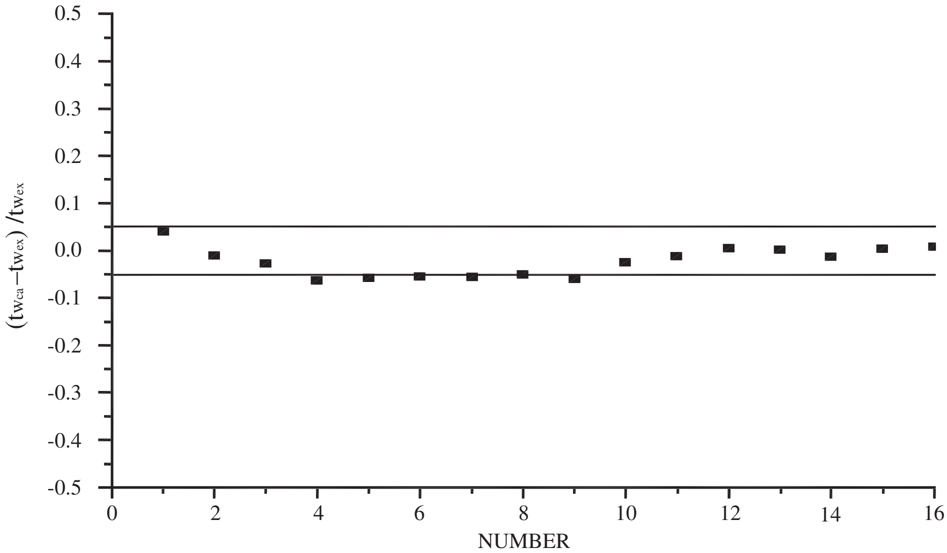



**Fig. 4** The tube wall temperature comparison of calculated and experimental data (sub-cooling water)

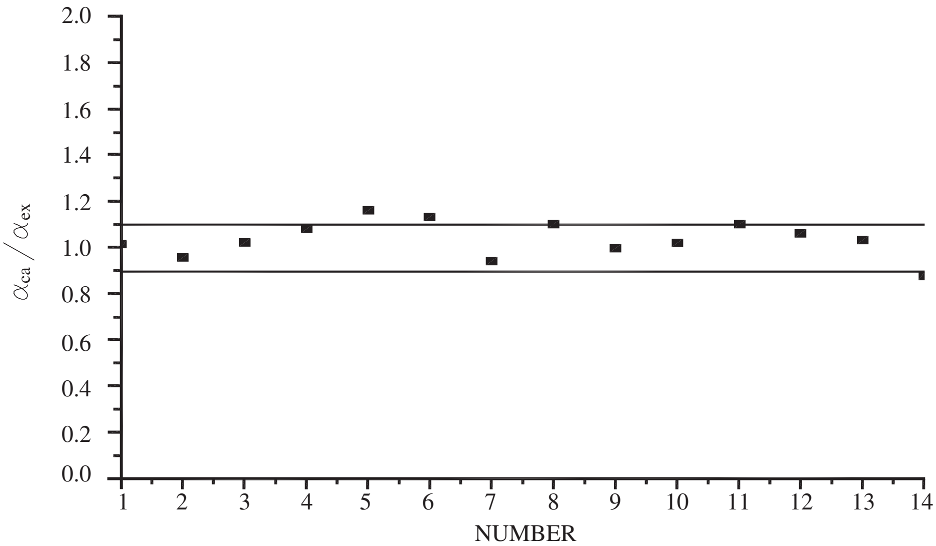



**Fig. 5** The tube wall temperature comparison of calculated and experimental data (super-heated steam)

**Table 2** Comparison of operation results and design parameters

| | Design parameter | Operational measured |
|---|---|---|
| Power (MW) | 10 | 10.2 |
| Pressure in the primary side (MPa) | 3.0 | 2.94 |
| Temperatures in the primary side (°C) | 250/700 | 236/699.2 |
| Flow rate in the secondary side (kg/s) | 3.49 | 3.56 |
| Pressure at the outlet of the secondary side (MPa) | 4.0 | 3.45 |
| Inlet temperatures of the secondary side (°C) | 104 | 99 |

er for 72 hours. Results of the operation and the comparison of operation results and design parameters are shown in **Table 2**.

It can be seen from Table 2, in case of that the practical power increased about 2% and temperature head decreased about 1%. That means the heat transfer surface still has a certain abundance.

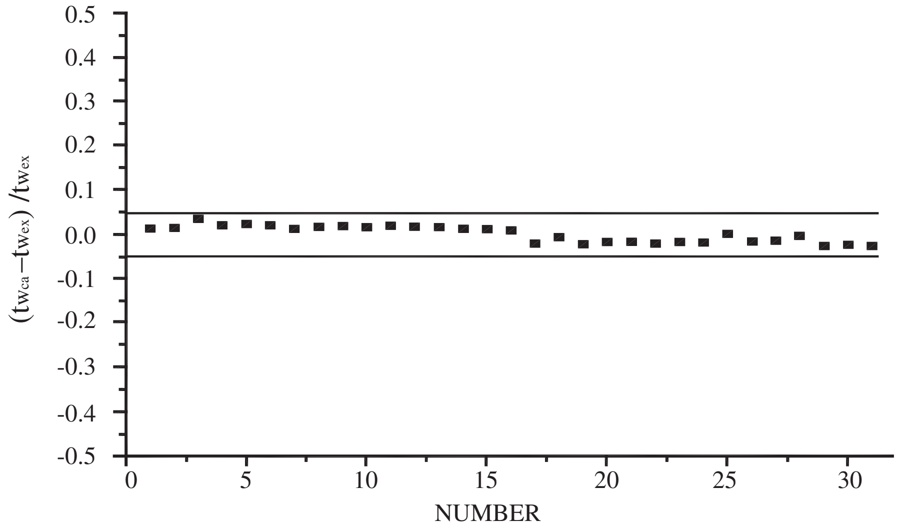



**Fig. 6** The tube wall temperature comparison of calculated and experimental data (the two-phase flow)

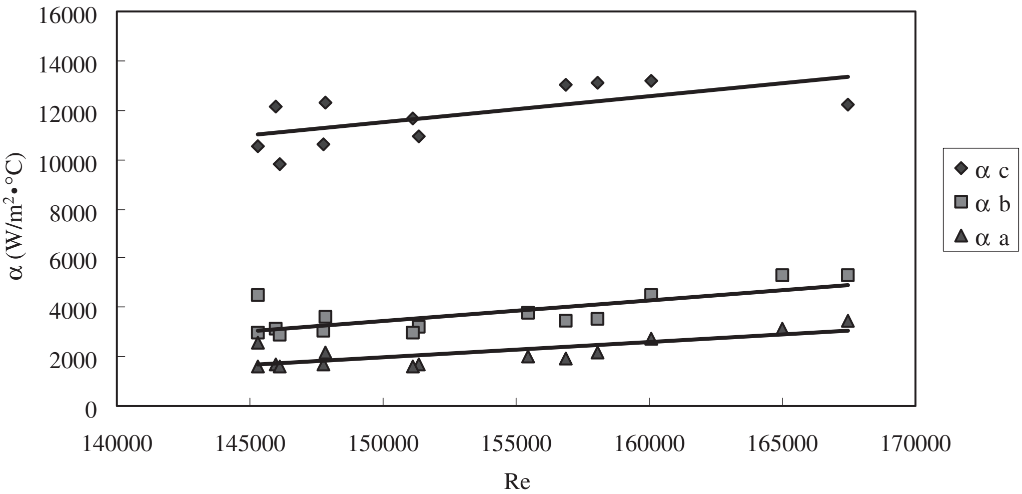



**Fig. 7** The experimental heat transfer coefficient for different steam contents and different positions

## IV. Verification of the Centrifugal Force Effect on the Heat Transfer For the Helical Tube

In the helical tube the flow is influenced by the gravitational force, centrifugal force and secondary flow. Therefore in the flow pattern and the distribution of the void fraction, there are some differences between the straight and helical tubes. Consequently, in the wall temperature distribution and heat transfer, the helical tube has its inherent thermal-hydraulic features.

In a helical tube, coolant flow is influenced by the gravitation force. It causes the steam–water separation in the vertical direction. When the velocity of the coolant flow is relatively low, the steam phase is concentrated on the upper part of the tube. At the same time, the influence of the centrifugal force causes the steam–water separation in the horizontal direction. It drives the steam phase concentrate on the inner part of the tube ring. Based on these factors, the wall temperature shows its maximum value between the top and the inner side of the tube ring.

In order to verify these effects, in the STGM-10 facility, 12 thermo-couples are arranged on the helical tube for 4 different sections. On every section there are 3 thermo-couples with a interval of 120° (on the top, inner and outer, see Fig. 3) to measure the tube wall temperature on the different positions for getting the heat transfer coefficient at the different position on the same tube section.

The experimental heat transfer coefficient for different void fraction and different positions are given in **Fig. 7**.

Where $\alpha_a$ is the heat transfer coefficient on the top, $\alpha_b$ is the inner position and $\alpha_c$ is the outer position. From Fig. 7 can be seen, in the helical tube steam content distribution is asymmetrical. Because of the higher void fraction and lower water flow velocity on the measured sections, the steam phase tends to the top and inner side of the tube ring. The maximum wall temperature appears on the point between the top and the inner side, where the heat transfer coefficient is the minimum of the section.

The asymmetric distribution of steam phase proved that in the helical tube, the tendency of steam–water separation is existed.

## V. Conclusions

This paper describes the study results for the heat transfer of the helical tube. From the study results following conclusions can be made:

From experimental and operation results, the design of the HTR-10 SG is successful. The operation parameters satisfied the design requirements.

(1) In the design of the HTR-10 SG, the heat transfer calculation was performed on the base of straight tube heat transfer calculation with an additional correction of the helical tube. Experimental results show that this method can be used for the HTR-10 SG design.

(2) Based on the test results, the experimental formulas of heat transfer coefficient of the He flow across the helical tube, and the heat transfer coefficient of water and the two-phase flow are obtained and proved that they can be used for HTR-10 SG design.

(3) Based on the test results, in the helical tube, the coolant flow is influenced by the gravitation force and the centrifugal force. The gravitational force causes the steam–water separation in the vertical direction. The centrifugal force causes the steam–water separation in the horizontal direction. So the asymmetric distribution of steam phase is existed in the helical tube. The minimum heat transfer coefficient appears on the point between the top and the inner side of the helical ring.

The successful operation of the HTR-10 SG presents some useful experiences for the heat transfer study of the small helical tube.

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